



University of Technology, Sydney

Faculty of Engineering and IT

**STUDY OF DRAG TORQUE IN A TWO-SPEED  
DUAL CLUTCH TRANSMISSION ELECTRIC  
VEHICLE POWERTRAIN SYSTEM**

A thesis submitted for the degree of

**Doctor of Philosophy**

**Xingxing Zhou**

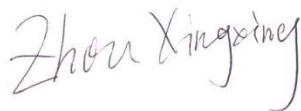
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## NOMENCLATURE

### Global abbreviations used in this thesis

DCT	=	dual clutch transmission
MT	=	manual transmission
AMT	=	automated manual transmission
CVT	=	continuously variable transmission
EV	=	electric vehicle
HEV	=	hybrid electric vehicle
EM	=	electric machine

### Chapter 3 Notation

TCU	=	transmission control unit
VCU	=	vehicle control unit
DCT	=	dual clutch transmission
$m_v$	=	vehicle mass (kg)
$\alpha$	=	vehicle acceleration ( $m/s^2$ )
$r_t$	=	tyre radius (m)
$T_{EM}$	=	electric machine torque (N.m)
$\gamma$	=	gear ratio
$T_v$	=	vehicle torque (N.m)
$C_R$	=	coefficient of rolling resistance
$g$	=	gravity ( $m/s^2$ )
$\phi$	=	road incline (Rad)
$C_D$	=	drag coefficient
$\rho$	=	air density ( $kg/m^3$ )
$A_v$	=	vehicle frontal area ( $m^2$ )
$V_v$	=	vehicle speed (m/s)
$P_R$	=	vehicle resisting power (W)
$\omega_v$	=	vehicle rotational speed (rad/s)

### Chapter 5 Notation

$A_g$	=	arrangement constant for gearing
$b_w$	=	face width in contact (mm)
$C1$	=	clutch 1
$C2$	=	clutch 2
$D$	=	outside diameter of the gear (mm)
$d_m$	=	diameter (mm)
$E_{DCT}$	=	efficiency of DCT
$F$	=	total face width (mm)

$F_b$	= applied load (N)
$f$	= turbulent flow coefficients
$f_0$	= bearing dip factor
$f_g$	= gear dip factor
$f_m$	= mesh coefficient of friction
$f_L$	= bearing load empirical factor
$H_s$	= sliding ratio at the start of approach action
$H_t$	= sliding ratio at the end of recess action
$h$	= tangential line velocity modifying exponent
$h_c$	= clutch plate's clearance (mm)
$h_{con}$	= length of concentric shaft (mm)
$j$	= viscosity modifying exponent
$K$	= load intensity ( $N/mm^2$ )
$L$	= length of the gear (mm)
$M$	= mesh mechanical advantage
$M_t$	= transverse tooth module
$N$	= number of frictional surface
$n_1$	= pinion rotational speed (rpm)
$n_{motor}$	= motor speed (rpm)
$P_B$	= power losses caused by bearings drag torque (KW)
$P_{Con}$	= power losses caused by concentric shaft drag torque (KW)
$P_{bl}$	= load independent power loss (KW)
$P_{bv}$	= speed independent power loss (KW)
$P_{Ch}$	= power losses caused by gear churning (KW)
$P_{Cl}$	= power losses caused by wet clutch plates drag torque (KW)
$P_G$	= power losses caused by gear meshing drag torque (KW)
$P_{GW}$	= individual gear windage and churning loss (KW)
$P_L$	= total power losses in DCT (KW)
$P_{oil}$	= oil seal power losses (KW)
$Q$	= flow rate ( $m^3/s$ )
$T_1$	= pinion torque (N.m)
$T_B$	= drag torque caused by bearings (N.m)
$T_{B(1,2)}$	= drag torque caused by bearing (1) and (2) (N.m)
$T_{Cl}$	= drag torque caused by wet clutch packs (N.m)
$T_{Ch}$	= drag torque caused by churning (N.m)
$T_{con}$	= drag torque caused by concentric shafts viscous shear resistance (N.m)
$T_{1st\_output}$	= output torque of the outer concentric shaft (N.m)
$T_{1st\_output}$	= output torque of the inner concentric shaft (N.m)
$T_{final\_output}$	= final output torque from DCT (N.m)
$T_{GM}$	= drag torque caused by gear pairs meshing (N.m)
$T_{GM_{1st\_pair}}$	= drag torque caused by 1 <sup>st</sup> gear pair meshing (N.m)
$T_m$	= motor output torque (N.m)
$V$	= pitch line velocity (m/s)
$V_{oil}$	= velocity for oil seal (rpm)
$R_{con\_i}$	= outer radius of the inner shaft (mm)
$R_{con\_o}$	= inner radius of the outer shaft (mm)
$R_f$	= roughness factor

$R_{o2}$	= gear outside radius (mm)
$R_{w2}$	= gear operating pitch radius (mm)
$R_{o1}$	= pinion outside radius (mm)
$R_{w1}$	= pinion operating pitch radius (mm)
$r$	= gear ratio
$r_m$	= mean radius (mm)
$r_o$	= outer radius of the clutch (mm)
$r_{1st}$	= 1 <sup>st</sup> gear ratio
$r_{2nd}$	= 2 <sup>nd</sup> gear ratio
$z_1$	= number of pinion teeth
$z_2$	= number of gear teeth
$\alpha_w$	= transverse operating pressure angle (°)
$\beta$	= generated helix angle (°)
$\beta_w$	= operating helix angle (°)
$\mu$	= viscosity of the oil ((Ns/m <sup>2</sup> ))
$\nu$	= kinematic oil viscosity (m <sup>2</sup> /s)
$\rho_{oil}$	= density of oil (kg/m <sup>3</sup> )
$\omega_{EM}$	= output speed of the electric motor (rpm)
$\Delta\omega$	= relative speed between two concentric shafts (rpm)
$\Delta\omega_{c1}$	= relative speed within clutch 1 (rpm)
$\Delta\omega_{c2}$	= relative speed within clutch 2 (rpm)
$\nabla p$	= pressure difference between the input and output of clutch pair (pa)

## Chapter 7 Notation

$A_{ca}$	= surface area of transmission case (m <sup>2</sup> )
$A_{oil}$	= oil-side surface area of transmission case (m <sup>2</sup> )
$c_{oil}$	= specific heat of oil (J/kg.K)
$c_{DCT}$	= the average specific heat of transmission (J/kg.K)
$h_{ca}$	= height of gearbox housing (m)
$G_r$	= grashoff number
$k$	= heat transmission coefficient (W/m <sup>2</sup> K)
$m_{oil}$	= mass of oil
$P_B$	= power losses caused by bearings drag torque (kW)
$P_{Con}$	= power losses caused by concentric shaft drag torque (kW)
$P_{Ch}$	= power losses caused by gear churning (kW)
$P_{Cl}$	= power losses caused by wet clutch plates drag torque (kW)
$P_G$	= power losses caused by gear meshing drag torque (kW)
$P_L$	= total power losses in DCT (kW)
$Q_{Ca}$	= quantity of heat dissipated by the transmission case (J)
$Q_{all}$	= overall quantity of DCT heat (J)
$Q_{oil}$	= quantity of heat absorbed by oil (J)
$Q_{DCT}$	= quantity of heat absorbed by DCT structure (J)
$Re$	= Reynold's number
$T_{oil}$	= oil temperature (Kelvin)

$T_{en}$	= environment temperature (Kelvin)
$T_0$	= the primary temperature (Kelvin)
$T_{wall}$	= temperature of housing wall (Kelvin)
$\alpha_{oil}$	= oil-side heat transfer coefficient (W/m <sup>2</sup> K)
$\alpha_{ca}$	= air-side heat conduction (W/m <sup>2</sup> K)
$\alpha_{con}$	= convection part(W/m <sup>2</sup> K)
$\alpha_{rad}$	= radiation part (W/m <sup>2</sup> K)
$\alpha_{free}$	= heat transfer coefficient due to free convection (W/m <sup>2</sup> K)
$\alpha_{forced}$	= heat transfer coefficient due to forced convection (W/m <sup>2</sup> K)
$\lambda_{wall}$	= coefficient of thermal conduction of housing (W/mK)
$\delta_{wall}$	= mean housing wall thickness (m)
$\epsilon$	= emission ratio
$\eta^*$	= temperature ratio
$v_{air}$	= impingement velocity (m/s)
$\Delta T$	= increased temperature caused by absorbing heat (Kelvin)

## Chapter 8 Notation

$A_V$	= vehicle frontal area( m <sup>2</sup> )
$C_D$	= drag coefficient
$C_R$	= coefficient of rolling resistance
$E_{DCT}$	= efficiency of DCT
$E_{motor}$	= efficiency of motor
$f_{obj}$	= objective function
$g$	= gravity (m.s <sup>2</sup> )
$m_V$	= vehicle mass (kg)
$N$	= number of surfaces of clutch plates
$P_R$	= vehicle resisting power (W)
$P_B$	= power losses caused by bearings drag torque (KW)
$P_{Con}$	= power losses caused by concentric shaft drag torque (KW)
$P_{Ch}$	= power losses caused by gear churning (KW)
$P_{Cl}$	= power losses caused by wet clutch plates drag torque (KW)
$P_G$	= power losses caused by gear meshing drag torque (KW)
$P_L$	= total power losses in DCT (KW)
$p_{max}$	= clutch plates maximum permissible pressure (MPa)
$r_t$	= tyre radius (m)
$r_i$	= inner radius of clutch plates (m)
$r_o$	= outer radius of clutch plates (m)
$T_{EM}$	= electric machine torque (N.m)
$T_V$	= vehicle torque (N.m)
$T_{max}$	= maximum torque transmitted by clutch (N.m)
$V_V$	= vehicle speed (m/s)
$V_{shift}$	= final shift speed point (km/h)
$V_{index}$	= index of original shift point (km/h)
$\gamma$	= gear ratio
$\gamma_{first}$	= the 1 <sup>st</sup> gear ratio



$\gamma_{\text{second}}$  = the 2<sup>nd</sup> gear ratio  
 $\alpha$  = vehicle acceleration ( $\text{m/s}^2$ )  
 $\mu$  = dynamic viscosity ( $\text{N/m}^2$ )  
 $\varphi$  = road incline (Rad)  
 $\omega_v$  = vehicle rotational speed (rad/s)  
 $\rho$  = air density ( $\text{kg/m}^3$ )

## **ABSTRACT**

Pure electric vehicles are widely looked as a potential avenue to reduce fossil fuel consumption and emission in the long term in the transportation section. Pure electric vehicles currently being used in the market are mainly equipped with single speed transmissions, with tradeoffs between dynamic (such as climbing ability, top speed, and acceleration) and economic performance (drive range). The employment of two-speed dual clutch transmission (DCT) in electric vehicles is likely to improve average motor efficiency and range, and even can reduce the required motor size. In order to comprehensively improve the electric vehicle powertrain system efficiency, it is necessary to fully consider the proposed two-speed transmission drag torque, including its influences and potential applications.

The focus of this thesis is on studying the drag torque within two-speed dual clutch transmissions. Different source of drag torque in the DCT are theoretically analysed and modelled, including torsional resistance caused by viscous shear caused between wet clutch plates and concentrically aligned shafts, gear mesh friction and windage, oil churning, and bearing losses. Then experimental works are carried out on UTS electric vehicle powertrain system test rig. Outcomes of experimentation on drag torque confirm that simulation results agree well with the test data. Then, based on the drag torque study, the thermal behaviour of the transmission is analysed via both theoretical and experimental investigation. Finally, integrated optimal design of electric vehicle powertrain system equipped with two-speed DCT is performed, considering drag torque, shift schedule, electric motor selection, gear ratio design and wet clutch design.